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HEAT TRANSFER IN A COMMERCIAL HEAT EXCHANGER

By

B. E. SHORT

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Engineering Research Series No. 29

Bureau of Engineering Research

Division of the Conservation and Development of the Natural Resources of Texas

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The benefits of education and of useful knowledge, generally diffused through a community, are essential to the preservation of a free government.

Sam Houston

Cultivated mind is the guardian genius of Democracy, and while guided and controlled by virtue, the noblest attribute of man. It is the only dictator that freemen acknowledge, and the only security which freemen desire.

Mirabeau B. Lamar

CONTENTS

HEAT TRANSFER IN A COMMERCIAL HEAT EXCHANGER

I. INTRODUCTION

Scope.—This bulletin presents the results of tests made on a horizontal shell-and-tube type heat exchanger having a heating surface of about twenty square feet. Heated fluids with viscosities ranging from 0.44 to 40 centipoises were pumped through the shell; water in the tubes served as the cooling medium. The overall coefficient of heat transfer was determined for several velocities of each fluid while the velocity of the cooling water remained constant.

Purpose.—The investigation was not undertaken for the purpose of establishing new theories or equations, but to determine the practicability of correlating the results of tests on a commercial (multiple-tube) heat exchanger with results obtained from laboratory data on single-tube experimental exchangers and, also, to establish a basis by which a rational analysis could be made of data obtained from tests on any heat exchanger.

Acknowledgments.—The tests were made in the laboratory of the Department of Mechanical Engineering in conjunction with the Bureau of Engineering Research, The University of Texas, Austin, Texas.

The authors are indebted to Professors H. E. Degler, A. Romberg, E. P. Schoch, and A. Vallance for their valuable suggestions relative to the preparation of the manuscript of this bulletin.

Theoretical considerations.—It is generally recognized that the rate of heat transfer from a fluid to the wall of the containing vessel varies:

directly with
- (1) the velocity
- (2) the heat capacity (specific heat)
- (3) the heat conductivity
- (4) the density

and inversely with

(1) the absolute viscosity of the fluid
(2) the diameter of the containing vessel.

This theory, based on mathematical dimensional analysis, has been substantiated by physical tests on laboratory apparatus. Newton's law implies that

$$\frac{Q}{t} = U\ A\ T \qquad (1)$$

where

Q = quantity of heat transferred in the time—(t)
U = constant for given conditions
A = transfer surface
T = average difference in temperature between the two substances.

The constant U is the "coefficient of heat transfer" which permits all of the variables named in the first paragraph to be considered in Newton's equation. This constant is also governed by the multiplicity of mediums through which the heat must flow. By maintaining constant conditions on one side of the containing wall, it is possible to show the variation in the "overall coefficient" with the different factors.

Summary.—In this bulletin the "overall coefficient" is shown to be a function of the velocity and viscosity and, since the friction loss is a function of these variables, the "overall coefficient" is shown to be a function of the pressure drop. The relation of the rate of heat transfer to the power required to move the fluid along the transfer surface is the fundamental factor in the design of such apparatus.

The effect of increasing the diameter of the passage and the effect of decreasing the length of the path through the exchanger is shown by the relation of the transfer rates for "single-pass" and "double-pass" flow for equal linear shell fluid velocities.

Shell fluid velocities between 0.30 and 5.5 feet per second were used in these tests, while the viscosities of the fluids ranged from 0.44 to 40 centipoises.

The resulting "overall coefficients" were approximately ten times as great for the least viscous fluid as for the most highly viscous fluid for the same linear velocity.

II. APPARATUS AND METHOD

The various tests were performed on a commercial horizontal shell-and-tube type heat exchanger having a heating surface of about twenty square feet. Figure 1 shows a diagrammatic sketch of the exchanger.

Description of exchanger.—A cylindrical one-quarter inch steel shell having an inside diameter of six inches encloses a bundle of thirty ⅝-inch tubes which are five feet one inch in length. The tubes terminate in tube sheets at each end. One tube sheet is held rigidly to the shell while the other terminates in what is known as a floating head.

The flow of cooling medium through the tubes is divided in two passes of fifteen tubes each by the baffle *K*. The flow of fluid through the shell is divided by a longitudinal baffle *J* at the center of the shell. To this longitudinal baffle are fastened narrow rings or semi-circular baffles *H;* these rings decrease the effective cross-sectional area of the shell fluid passage, and result in a flow practically parallel to the length of the tubes.

Explanation of flow of fluids.—The piping, Fig. 2, to the two-inch inlet and outlet nozzles of the shell passage was so arranged as to allow flow to take place in what is termed in this bulletin as "single-pass" or "double-pass." For double-pass flow the fluid entered the shell at nozzle *E* and was discharged at nozzle *D*. For the single-pass flow the fluid was allowed to enter the shell through nozzles *E* and *D* simultaneously and was discharged from the shell through a single nozzle *B*. It may be noted that the cross-sectional area of the shell passage in single-pass was twice that for double-pass flow, but the length of the path of flow of the shell fluid in single-pass was but one-half that for the double-pass arrangement.

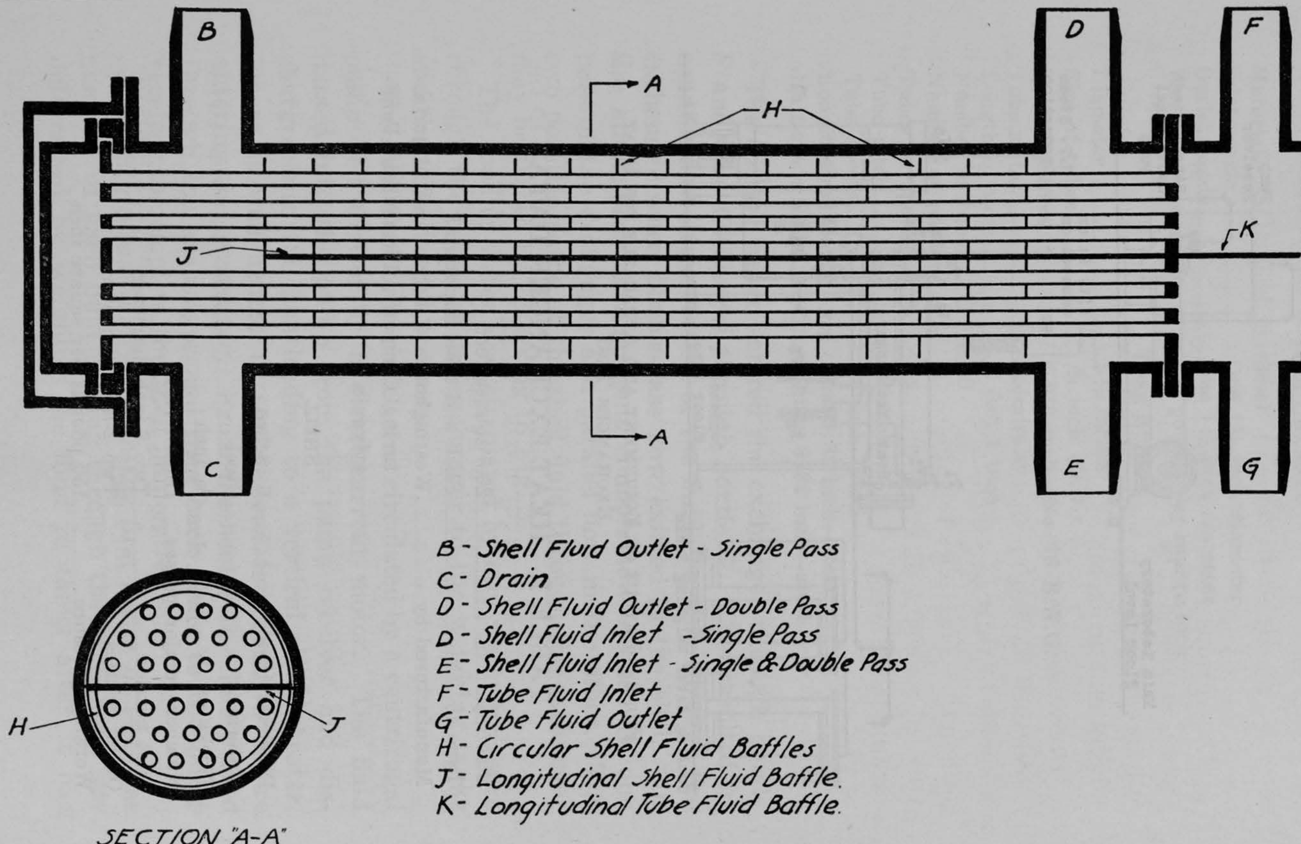

FIG. 1. SKETCH OF HEAT EXCHANGER

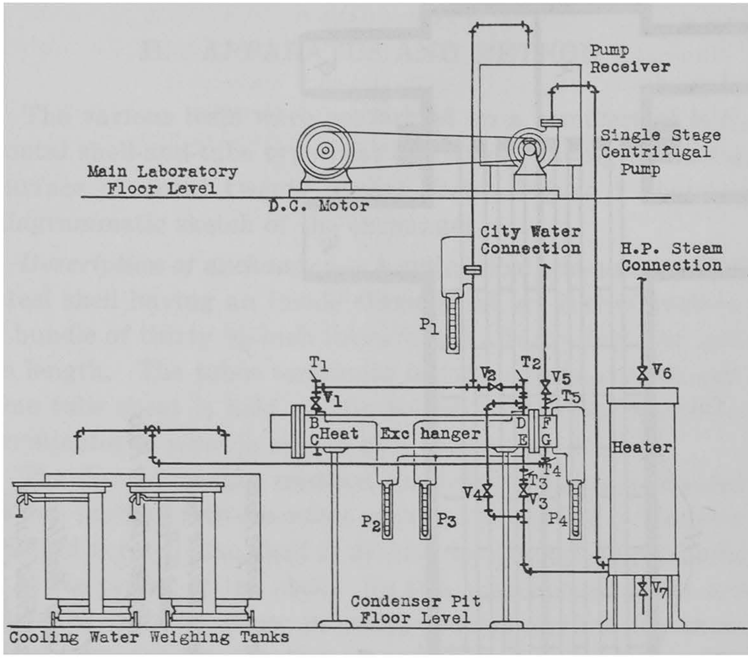

FIG. 2. LAYOUT OF EQUIPMENT AND APPARATUS FOR HEAT EXCHANGER TEST

TABLE I. HEAT EXCHANGER DATA

Size	20.5 square feet
Type	Shell and tube, horizontal
Manufactured by	Westinghouse Electric and Manufacturing Company, Pittsburgh, Pennsylvania

SHELL

Diameter inside	6 inches
Material	Steel
Thickness	¼ inch
Number—major passes	One or two
Inlet nozzles	Two 2-inch diameter
Outlet nozzles	Two 2-inch diameter
Working pressure	150 pounds per square inch

TUBE CHANNEL

Material	Steel
Inlet nozzle	One 1½-inch diameter
Outlet nozzle	One 1½-inch diameter
Working pressure	100 pounds per square inch

TUBE BUNDLE

Diameter	5½ inches
Size of tubes	⅝ inch (O.D.)
Tube thickness	0.049 inch—No. 18 B.W.G.
Tube metal	Admiralty
Length of tube	5 feet 1 inch
Number of tubes	30
Number of passes	2
Tubes per pass	15
Tube pitch	¾ inch
Tube sheets	½ inch—steel
Longitudinal baffle	One 3/32 inch—steel
Half-circle baffles	Thirty 3/32 inch—steel

The cooling water entered the exchanger through nozzle *F* and was discharged through nozzle *G*. This resulted in counter-flow arrangement of the fluids passing through the exchanger when double-pass flow existed in the shell. In the single-pass arrangement counter-flow existed in one pass of the exchanger and parallel flow in the other. The two possible paths of the shell fluid through the exchanger may be traced by referring to Fig. 2.

The cooling water was circulated by city water pressure through the tubes of the exchanger to the weighing tanks and then discharged to waste.

The shell fluid to be cooled was circulated by a centrifugal pump belt driven by a direct-current motor. The fluid passed into the pump from the pump receiver and discharged into the line leading to a vertical closed heater, where it was heated to the desired temperature before entering the exchanger. From the heater the fluid passed through the exchanger and then back again to the pump receiver to complete the shell fluid cycle.

Quantity measurements.—In the first series of tests the quantity of shell fluid passing through the exchanger was determined by weighing the fluid in tanks after it had passed through the exchanger. In order to facilitate a greater range of quantities of flow, the shell fluid was metered by use of an orifice in the line as shown in Fig. 2. The orifice was calibrated in place by direct weighing of the fluid passing through the orifice at different orifice pressure drops. The calibration curve for the orifice is shown by Fig. 3.

Fluid velocity regulation.—The velocity of the shell fluid passing through the exchanger was controlled by varying the speed of the pump and by adjusting the valves which

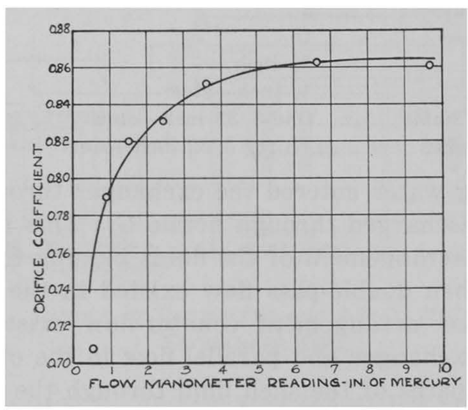

FIG. 3. ORIFICE CALIBRATION CURVE

controlled the discharge from the exchanger. The velocity of the tube fluid was varied by regulating the inlet valve V_5 to the tube passage.

Temperature measurements.—The temperatures were measured by long-stem mercurial thermometers which were read at least every five minutes. The thermometer wells, the positions of which are indicated on Fig. 2, were placed in ells or tees so as to assure a condition of turbulence about the stem of the wells.

Pressure measurements.—The pressure drop across a single-pass or double-pass of the exchanger and across the orifice used for metering the shell fluid was measured by the use of mercury-filled U-tube manometers as indicated by Fig. 2. The manometers were read at the same intervals as the thermometers. Manometer P_2 was used to measure the pressure drop across the exchanger when single-pass flow existed in the shell, while manometer P_3 measured the pressure drop when double-flow existed in the shell. Manometer P_4 was used to determine the pressure drop across the tube passage and particularly to check the uniformity of flow of the cooling water.

Time of tests.—The time of each test varied from fifteen to thirty minutes, and was determined by the constancy of the indications of the apparatus.

As may be noted from the data and result sheets, the fluid being cooled (the shell fluid) was first heated, in each case, to a predetermined temperature which was kept as nearly constant as possible after conditions of thermal equilibrium were established for each test. The amount of time necessary to establish thermal equilibrium varied and depended upon the changes made and upon the steam pressure in the line. There was no control of the incoming temperature of the cooling water; however, this temperature did not vary appreciably.

Shell fluid samples.—Samples of each oil used were taken shortly after the run, and determinations of the specific gravity and viscosity were made of each sample (except for the viscosity of the kerosene[1]) by the use of a hydrometer and a Saybolt Universal viscosimeter.

[1]The viscosity of the kerosene was obtained from a temperature-fluidity chart given in Hamor and Padgett's "Examination of Petroleum Oils," Fig. 38, page 85.

III. CALCULATIONS AND DATA

Series I of the tests was performed to study the relation of the velocity of the shell fluid to the overall heat transfer coefficient. Series II of the tests was performed to study the pressure loss of the shell fluid and its relation to the overall heat transfer coefficient in addition to similar data taken in series I of the tests.

Overall heat transfer coefficient.—The overall heat transfer coefficients were computed from the following equation:

$$Q = U\ A\ T \qquad (2)$$

where

Q = heat transferred in B.t.u. per hour

U = overall heat transfer coefficient in B.t.u. per hour per square foot per degree Fahrenheit mean temperature difference

A = transfer surface in square feet

T = mean temperature difference in degrees Fahrenheit.

The overall transfer coefficient was computed on the basis of the tube surface, which has an area of 20.5 square feet.

The heat passing through the transfer surface was taken as that absorbed by the cooling water. This value was checked by the heat given up by the fluid being cooled. Radiation from the shell was not considered, since this portion of the heat did not pass across the transfer surface.

Specific heat.—In the case of oil, the amount of heat given up or absorbed was calculated by using a value of specific heat determined from Fortsch and Whitman's equation.[2]

[2]Morris and Whitman, "Heat Transfer for Oils and Water in Pipes," Industrial and Engineering Chemistry Vol. 20 p. 236.

$$C = \frac{(T + 670)\ (2.10 - S)}{2030} \qquad (3)$$

where

C = specific heat

T = temperature in degrees Fahrenheit
S = specific gravity of the oil at 60 degrees Fahrenheit.

Mean temperature difference.—The arithmetic mean temperature difference was used in all computations where the error involved was less than 0.1 per cent.

Properties of oils.—Designations of the oils with the gravities at standard conditions and absolute viscosities at average conditions are given in Table II.

TABLE II

Properties of Oils Used as Shell Fluid

Oil	Degrees Baume 60°/60° F.	Specific Gravity 60°/60° F.	Absolute Viscosity at Average Temperature in Centipoises
A	38.0	0.835	0.90
B	24.7	0.906	4.65
C	19.6	0.936	8.30
D	18.8	0.941	11.60
E	27.8	0.888	10.40
F	23.8	0.911	5.60
G	41.4	0.814	0.90

The variation for the different oils of the relative viscosity (seconds Saybolt Universal) with temperature is shown by the curves in Fig. 4.

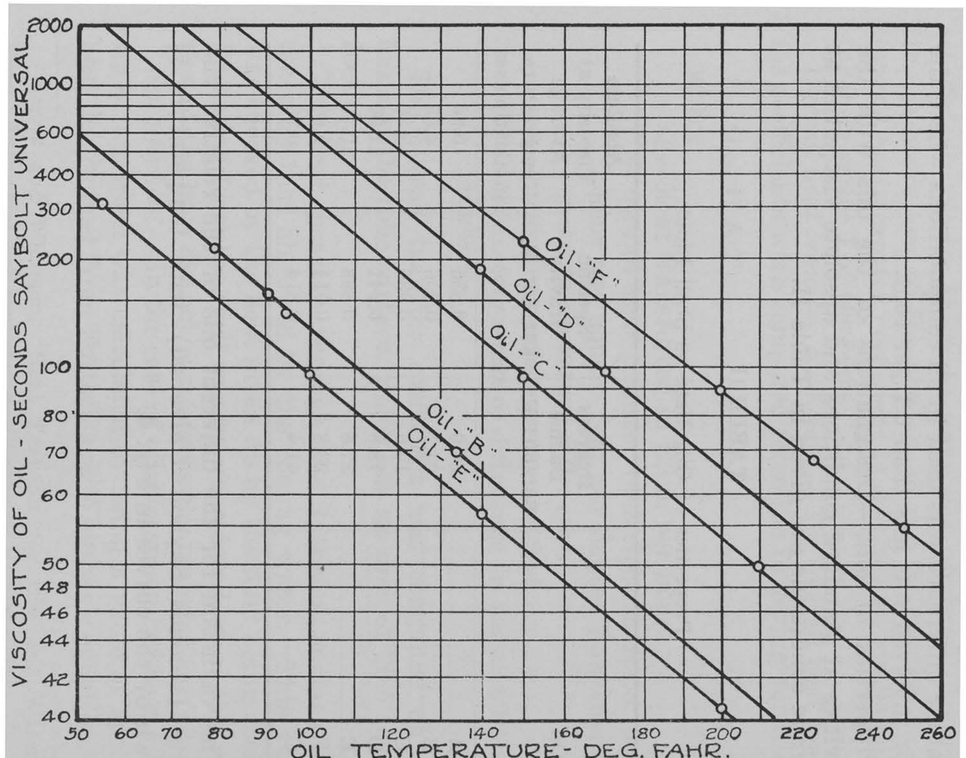

Fig. 4. Relation of Oil Viscosity to Temperature

TABLE III DATA AND RESULTS WATER COOLING WATER SERIES II.

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TUBE FLUID -OUT- DEG. F.	SHELL FLUID -IN- DEG. F.	SHELL FLUID -OUT- DEG. F.	MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEF-FICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
1	65.0	95.8	148.9	128.8	58.4	0.95	0.79	0.196	1.102	0.542	0.0358	0.0538	8,030	12,080	1.58	1.40	207.	
2	65.0	99.0	149.9	132.0	59.0	1.35	0.81	0.196	1.50	0.738	0.0349	0.0658	7,842	14,780	1.54	1.72	222.	
3	65.0	100.3	148.0	132.0	57.1	1.90	0.83	0.196	1.85	0.908	0.0351	0.0798	7,892	17,920	1.55	2.08	239.	
4	65.0	103.5	150.0	135.5	57.7	2.45	0.84	0.196	2.28	1.12	0.0349	0.0918	7,848	20,620	1.54	2.39	255.	
5	65.0	105.0	148.0	136.0	56.0	3.69	0.85	0.196	3.11	1.53	0.0349	0.1140	7,832	25,600	1.54	2.95	276.	Tests 1 to 9 incl. Double Pass Shell Fluid Velocity Variable
6	65.0	108.0	150.0	138.0	56.1	4.65	0.855	0.196	3.70	1.82	0.0350	0.1286	7,864	28,850	1.54	3.35	295.	
7	65.0	110.0	150.0	140.0	55.8	6.45	0.86	0.196	4.85	2.38	0.0352	0.1524	7,900	34,200	1.55	3.97	312.	
8	65.0	111.9	151.3	142.0	56.2	8.48	0.86	0.196	5.82	2.86	0.0352	0.1750	7,912	39,300	1.55	4.56	323.	
9	65.0	112.3	149.6	140.5	54.4	10.70	0.86	0.196	7.05	3.46	0.0354	0.1968	7,952	44,200	1.56	5.12	339.	
10	65.0	111.6	150.0	140.0	54.7	8.17	0.86	0.196	3.28	1.61	0.0352	0.1721	7,904	38,700	1.55	2.24	329.	
11	65.0	112.4	151.0	141.0	55.3	6.85	0.86	0.196	2.66	1.31	0.0331	0.1510	7,446	33,900	1.46	1.96	312.	Tube Fluid - Water Shell Fluid - Water Abs. Vis. - 0.44 c.p. at Average Temp.
12	65.0	112.1	151.6	140.4	55.6	5.80	0.86	0.196	2.50	1.23	0.0319	0.1448	7,168	32,500	1.40	1.88	297.	
13	65.0	110.1	150.5	138.6	55.4	4.25	0.853	0.196	2.03	0.998	0.0321	0.1228	7,204	27,600	1.41	1.60	286.	
14	65.3	106.3	149.4	135.6	55.6	3.10	0.846	0.196	1.66	0.815	0.0332	0.1041	7,456	23,400	1.46	1.35	269.	
15	65.5	103.1	149.1	133.0	56.0	1.85	0.827	0.196	1.25	0.613	0.0347	0.0786	7,812	17,660	1.53	1.02	255.	
16	66.0	98.6	148.4	128.9	56.0	1.06	0.793	0.196	0.91	0.447	0.0369	0.0569	8,288	12,800	1.62	0.74	236	
17	66.0	109.7	149.2	138.8	54.5	5.95	0.86	0.206	2.20	1.08	0.0354	0.1465	7,962	32,900	1.54	1.91	313.	
18	65.7	109.4	150.1	138.6	55.3	6.05	0.86	0.206	2.26	1.11	0.0367	0.1476	8,260	33,100	1.61	1.92	319.	
19	65.3	106.0	150.0	136.5	56.5	4.30	0.853	0.245	1.63	0.801	0.0396	0.1250	8,910	28,100	1.74	1.63	314.	Tests 10 to 28 incl. Single Pass Shell Fluid Velocity Variable
20	65.5	114.2	153.3	143.3	56.3	8.10	0.86	0.196	2.72	1.34	0.0348	0.1711	7,818	38,400	1.53	2.23	333.	
21	67.0	114.1	150.2	141.7	53.2	9.97	0.86	0.221	2.94	1.44	0.0355	0.1899	7,986	42,600	1.56	2.47	346.	
22	67.0	113.3	150.0	140.0	52.9	8.71	0.86	0.221	2.60	1.276	0.0360	0.1775	8,100	39,900	1.58	2.31	346.	
23	67.0	113.2	149.9	139.3	52.6	7.11	0.86	0.211	2.275	1.12	0.0352	0.1600	7,904	35,900	1.55	2.08	339.	
24	67.0	113.0	150.0	139.0	52.7	5.70	0.86	0.196	1.90	0.933	0.0355	0.1435	7,980	32,200	1.56	1.87	340.	
25	67.2	110.1	149.1	136.7	52.9	4.19	0.853	0.196	1.37	0.672	0.0352	0.1220	7,905	27,400	1.55	1.59	313.	
26	67.3	108.6	150.3	136.0	54.0	2.94	0.845	0.196	1.00	0.491	0.0354	0.1011	7,956	22,700	1.56	1.32	297.	
27	67.2	105.7	150.5	133.8	55.0	1.89	0.828	0.196	0.60	0.295	0.0355	0.0794	7,980	17,820	1.56	1.03	273.	
28	67.5	113.1	151.1	139.8	53.4	6.00	0.86	0.196	1.80	0.883	0.0352	0.1471	7,912	33,050	1.55	1.92	333.	

TABLE IV DATA AND RESULTS WATER COOLING OIL "A" SERIES II.

TEST NUMBER	TEMPERATURES					FLOW	ORIFICE	PRESSURE DROP			FLUID FLOWING				FLUID VELOCITY		OVERALL	REMARKS
	TUBE FLUID -IN- DEG. F.	TUBE FLUID -OUT- DEG. F.	SHELL FLUID -IN- DEG. F.	SHELL FLUID -OUT- DEG. F.	MEAN TEMP. DIFF. DEG. F.	MANO. READING INCHES HG.	COEF-FICIENT	TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	TUBE FT. PER SEC.	SHELL FT. PER SEC.	TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	
29	58.0	86.0	151.8	140.7	74.3	10.252	0.86	0.196	4.10	2.01	0.0348	0.2146	7,832	38,700	1.53	5.59	144.5	Tests 29 to 38 incl. Double Pass Shell Fluid Velocity Variable
30	58.0	85.6	152.8	140.6	74.9	8.752	0.86	0.196	3.63	1.78	0.0351	0.1980	7,900	35,750	1.55	5.16	142.5	
31	58.4	84.6	151.8	139.8	74.3	7.652	0.86	0.196	3.16	1.55	0.0351	0.1854	7,880	33,460	1.55	4.83	136.5	
32	58.6	83.9	151.3	138.3	73.6	6.602	0.86	0.196	2.75	1.35	0.0350	0.1719	7,866	31,020	1.54	4.48	132.0	
33	58.5	81.9	150.0	137.1	73.4	5.352	0.86	0.196	2.20	1.08	0.0352	0.1550	7,912	28,000	1.55	4.04	123.5	
34	58.5	80.0	152.1	137.6	75.6	3.650	0.85	0.196	1.63	0.80	0.0352	0.1265	7,908	22,830	1.55	3.29	110.0	
35	58.4	77.9	152.0	136.3	76.0	2.614	0.84	0.196	1.29	0.63	0.0352	0.1059	7,908	19,120	1.55	2.76	99.8	
36	58.0	74.7	151.6	134.0	76.4	1.502	0.82	0.196	0.85	0.42	0.0351	0.0784	7,880	14,150	1.55	2.05	85.4	
37	58.0	71.9	152.0	132.0	77.1	0.767	0.81	0.196	0.50	0.25	0.0351	0.0552	7,900	9,950	1.55	1.44	69.4	
38	58.0	69.8	159.3	133.7	82.6	0.402	0.80	0.196	0.30	0.15	0.0351	0.0390	7,896	7,030	1.55	1.02	55.1	
39	59.0	89.3	152.0	140.0	71.8	12.020	0.86	0.196	3.50	1.72	0.0350	0.2320	7,890	41,900	1.54	3.02	162.5	Tests 39 to 45 incl. Single Pass Shell Fluid Velocity Variable
40	59.1	88.4	150.9	139.1	71.3	9.890	0.86	0.196	3.01	1.48	0.0350	0.2106	7,888	38,050	1.54	2.74	159.0	
41	59.5	87.5	151.5	139.0	71.8	8.602	0.86	0.196	2.64	1.30	0.0352	0.1964	7,928	35,450	1.55	2.56	151.0	
42	59.5	86.3	151.4	138.3	71.9	7.402	0.86	0.196	2.36	1.16	0.0354	0.1822	7,956	32,900	1.56	2.37	145.0	
43	59.7	84.3	153.0	138.2	73.6	5.064	0.86	0.196	1.71	0.84	0.0354	0.1509	7,968	27,250	1.56	1.96	130.0	
44	59.7	80.4	150.4	134.7	72.5	3.064	0.85	0.196	1.16	0.57	0.0355	0.1160	7,972	20,920	1.57	1.51	111.2	
45	59.7	77.0	150.4	132.1	72.9	1.602	0.82	0.196	0.76	0.37	0.0353	0.0809	7,932	14,610	1.55	1.05	92.0	

Tube Fluid - Water
Shell Fluid - Oil "A"
Sp. Gr. - 0.804 at Average Temp.
Abs. Vis. - 0.9 c.p. at Average Temp.

TABLE V. DATA AND RESULTS WATER COOLING OIL 'B' SERIES II.

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TEMPERATURES: TUBE FLUID -OUT- DEG. F.	TEMPERATURES: SHELL FLUID -IN- DEG. F.	TEMPERATURES: SHELL FLUID -OUT- DEG. F.	TEMPERATURES: MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEF-FICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	PRESSURE DROP: SHELL INCHES HG.	PRESSURE DROP: SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	FLUID FLOWING: SHELL CU. FT. PER SEC.	FLUID FLOWING: TUBE LB. PER HOUR	FLUID FLOWING: SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	FLUID VELOCITY: SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
46	59.5	85.9	199.7	186.8	120.5	7.457	0.86	0.196	2.79	1.37	0.0352	0.1777	7,916	33,900	1.55	2.31	84.8	Tests 46 to 51 incl. Single Pass Shell Fluid Velocity Variable
47	59.5	84.8	200.0	186.7	121.2	6.320	0.86	0.196	2.30	1.13	0.0353	0.1632	7,936	31,100	1.56	2.13	81.0	
48	59.7	83.6	200.6	187.1	122.2	5.195	0.86	0.196	1.87	0.92	0.0352	0.1480	7,920	28,220	1.55	1.93	75.7	
49	60.0	81.3	200.4	186.4	122.8	3.982	0.85	0.196	1.40	0.69	0.0358	0.1284	8,048	24,500	1.58	1.67	68.3	
50	60.0	79.5	200.9	185.6	123.5	2.520	0.84	0.196	1.10	0.54	0.0348	0.1010	7,816	19,260	1.53	1.32	60.4	
51	59.5	71.6	198.0	178.0	122.4	0.732	0.80	0.196	0.57	0.28	0.0364	0.0518	8,184	9,870	1.61	0.67	39.5	
52	59.0	83.6	199.5	188.0	122.4	7.982	0.86	0.196	3.90	1.92	0.0349	0.1836	7,848	35,000	1.54	4.78	77.0	Tests 52 to 61 incl. Double Pass Shell Fluid Velocity Variable
53	58.6	82.0	198.8	186.8	122.5	7.265	0.86	0.196	3.46	1.70	0.0355	0.1748	7,974	33,350	1.57	4.55	74.5	
54	58.0	82.0	200.0	188.0	124.0	6.232	0.86	0.196	3.15	1.55	0.0348	0.1620	7,828	30,900	1.53	4.21	74.1	
55	58.0	79.9	200.0	188.0	125.1	5.182	0.86	0.196	2.71	1.33	0.0348	0.1479	7,816	28,220	1.53	3.85	66.8	
56	58.0	78.1	200.5	189.3	126.8	3.882	0.85	0.196	2.14	1.05	0.0350	0.1269	7,860	24,220	1.54	3.30	61.0	
57	58.0	75.0	200.4	186.3	126.8	2.782	0.84	0.196	1.70	0.84	0.0355	0.1060	7,990	20,240	1.57	2.76	53.1	
58	58.0	73.0	200.9	186.0	128.0	1.707	0.83	0.196	1.10	0.54	0.0354	0.0821	7,940	15,680	1.56	2.14	45.4	
59	58.0	69.0	198.2	180.6	125.9	0.807	0.80	0.196	0.60	0.30	0.0354	0.0545	7,940	10,420	1.56	1.42	33.9	
60	58.5	69.5	201.7	184.5	128.9	0.832	0.80	0.196	0.60	0.29	0.0346	0.0553	7,776	10,550	1.53	1.44	33.6	
61	58.6	73.4	199.5	184.1	125.8	1.782	0.83	0.196	1.15	0.57	0.0348	0.0840	7,824	16,020	1.53	2.19	45.2	
62	58.7	67.1	197.5	167.7	119.7	0.232	0.79	0.196	0.35	0.17	0.0351	0.0288	7,872	5,490	1.55	0.38	27.0	Test 62-Single Pass Shell Fluid Velocity Variable
63	58.6	67.9	201.3	184.5	129.7	0.532	0.80	0.196	0.43	0.21	0.0355	0.0443	7,980	8,450	1.57	1.15	28.0	Test 63 and 64 Double Pass Shell Fluid Velocity Variable
64	58.8	62.5	202.0	172.8	126.7	0.232	0.79	0.196	0.20	0.10	0.0355	0.0288	7,980	5,490	1.57	0.75	11.3	

Tube Fluid - Water
Shell Fluid - Oil "B"
Sp. Gr. - 0.85 at Average Temp.
Abs. Vis. - 4.65 c.p. at Average Temp.

TABLE VI DATA AND RESULTS WATER COOLING OIL "C" SERIES II.

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG.F.	TEMPERATURES: TUBE FLUID -OUT- DEG.F.	TEMPERATURES: SHELL FLUID -IN- DEG.F	TEMPERATURES: SHELL FLUID -OUT- DEG.F.	TEMPERATURES: MEAN TEMP. DIFF. DEG.F.	FLOW MANO. READING INCHES HG.	ORIFICE COEF-FICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	PRESSURE DROP: SHELL INCHES HG.	PRESSURE DROP: SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	FLUID FLOWING: SHELL CU. FT. PER SEC.	FLUID FLOWING: TUBE LB. PER HOUR	FLUID FLOWING: SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	FLUID VELOCITY: SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR/SQ.FT PER °F. M.T.D.	REMARKS
65	67.5	83.2	200.0	190.0	119.7	4.22	0.894	0.196	2.10	1.030	0.0345	0.1360	7,756	27,100	1.51	3.54	49.6	
66	67.5	82.8	200.0	190.0	119.8	3.70	0.892	0.196	1.93	0.950	0.0355	0.1269	7,972	25,250	1.56	3.30	49.7	
67	67.5	81.7	199.3	189.3	119.7	3.04	0.885	0.196	1.53	0.748	0.0347	0.1150	7,808	22,910	1.52	2.99	45.2	
68	67.4	80.5	199.3	189.1	120.2	2.45	0.883	0.196	1.25	0.614	0.0343	0.1023	7,736	20,400	1.50	2.66	41.2	
69	67.5	78.9	198.8	187.9	120.1	1.64	0.860	0.196	0.89	0.437	0.0347	0.0822	7,808	16,380	1.52	2.14	36.4	
70	67.5	77.0	200.0	187.5	121.5	0.97	0.833	0.196	0.60	0.295	0.0347	0.0612	7,812	12,200	1.52	1.59	29.9	Tests 65 to 77 incl. Double Pass Shell Fluid Velocity Variable
71	65.2	83.4	203.6	193.4	124.2	5.57	0.894	0.196	2.89	1.420	0.0352	0.1562	7,924	31,100	1.54	4.07	56.7	
72	65.0	81.9	199.4	189.8	121.2	4.80	0.894	0.196	2.50	1.228	0.0345	0.1450	7,768	28,900	1.51	3.77	53.0	
73	65.2	81.3	200.3	190.1	121.9	4.14	0.894	0.196	2.18	1.070	0.0350	0.1346	7,880	26,800	1.53	3.50	50.9	
74	65.2	78.9	201.5	190.2	123.8	2.49	0.883	0.196	1.45	0.711	0.0352	0.1032	7,928	20,540	1.54	2.69	43.1	
75	65.5	75.8	199.8	187.5	123.0	1.30	0.860	0.196	0.80	0.393	0.0352	0.0732	7,916	14,570	1.54	1.91	32.6	Tube Fluid - Water
76	65.5	74.8	199.1	186.0	122.4	0.90	0.833	0.196	0.60	0.295	0.0352	0.0590	7,912	11,730	1.54	1.54	29.3	Shell Fluid - Oil 'C'
77	65.3	73.0	200.0	186.0	123.9	0.55	0.833	0.196	0.35	0.172	0.0351	0.0461	7,904	9,180	1.54	1.20	24.0	Sp.Gr. - 0.886 at Average Temp.
78	65.5	84.0	201.1	190.5	121.1	5.40	0.894	0.196	2.25	1.105	0.0354	0.1539	7,960	30,600	1.55	2.00	56.6	Abs. Vis. - 8.3 c.p. at Average Temp.
79	65.5	83.0	200.0	190.0	120.8	4.60	0.894	0.196	2.00	0.982	0.0354	0.1420	7,980	28,260	1.55	1.85	59.5	
80	65.5	82.0	200.0	190.0	121.8	4.23	0.894	0.196	1.81	0.889	0.0354	0.1361	7,954	27,100	1.55	1.77	52.7	
81	65.3	79.0	200.5	189.5	122.8	2.15	0.883	0.196	1.35	0.661	0.0351	0.0958	7,904	19,100	1.54	1.25	43.0	
82	65.2	76.1	198.9	186.0	121.8	1.08	0.860	0.196	0.95	0.467	0.0354	0.0665	7,956	13,250	1.55	0.87	34.6	
83	65.1	70.2	198.4	175.0	119.1	0.10	0.805	0.196	0.35	0.172	0.0354	0.0190	7,964	3,770	1.55	0.25	16.9	Tests 78 to 88 incl. Single Pass Shell Fluid Velocity Variable
84	65.0	75.4	201.2	186.8	123.8	0.80	0.833	0.196	0.65	0.319	0.0354	0.0556	7,960	11,060	1.55	0.72	32.6	
85	65.0	77.1	200.5	187.7	123.1	1.30	0.860	0.196	0.90	0.443	0.0354	0.0732	7,956	14,580	1.55	0.95	38.3	
86	65.0	79.2	200.7	188.7	122.6	2.20	0.883	0.196	1.28	0.630	0.0355	0.0972	7,974	19,330	1.55	1.27	45.3	
87	64.9	80.9	200.9	190.2	122.7	3.52	0.883	0.196	1.78	0.874	0.0365	0.1226	8,228	24,350	1.60	1.60	52.6	
88	64.8	82.2	200.8	190.1	121.9	4.60	0.894	0.196	2.18	1.070	0.0354	0.1420	7,968	28,200	1.55	1.85	55.5	

TABLE VII DATA AND RESULTS WATER COOLING OIL "D" SERIES II

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TUBE FLUID -OUT- DEG. F.	SHELL FLUID -IN- DEG. F.	SHELL FLUID -OUT- DEG. F.	MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEF-FICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
89	67.2	83.9	200.4	192.6	120.9	6.797	0.903	0.196	4.38	2.15	0.0352	0.1742	7,904	34,900	1.54	4.53	53.6	
90	67.2	83.0	200.0	192.0	120.9	6.067	0.903	0.196	4.01	1.97	0.0354	0.1646	7,972	33,000	1.55	4.28	51.3	
91	67.0	82.2	200.4	192.3	121.7	4.917	0.903	0.196	3.40	1.67	0.0348	0.1482	7,844	29,700	1.53	3.86	47.8	
92	67.0	80.3	200.1	192.0	122.4	3.967	0.894	0.196	2.96	1.45	0.0358	0.1318	8,050	26,400	1.57	3.43	42.8	Tests 89 to 95 incl. Double Pass Shell Fluid Velocity Variable
93	67.0	79.2	200.0	191.0	122.4	2.630	0.883	0.196	2.26	1.11	0.0340	0.1061	7,656	21,250	1.49	2.76	37.4	
94	67.0	75.7	200.9	191.3	124.7	1.267	0.848	0.196	1.30	0.64	0.0352	0.0707	7,924	14,170	1.54	1.84	27.0	
95	67.0	72.9	199.1	188.1	123.6	0.567	0.833	0.196	0.66	0.324	0.0352	0.0465	7,920	9,320	1.54	1.21	18.5	
96	67.1	85.7	201.6	196.4	122.6	6.317	0.903	0.196	3.07	1.506	0.0352	0.1682	7,932	33,700	1.54	2.19	58.9	
97	67.3	84.8	199.4	195.0	121.2	5.717	0.903	0.196	2.80	1.373	0.0352	0.1663	7,940	33,300	1.54	2.17	56.0	
98	67.5	85.2	201.3	192.9	120.7	5.047	0.903	0.196	2.60	1.277	0.0345	0.1504	7,771	30,100	1.52	1.96	55.7	
99	67.5	82.8	200.5	191.8	121.0	3.317	0.894	0.196	1.83	0.898	0.0352	0.1205	7,924	24,100	1.54	1.57	49.0	Tests 96 to 102 incl. Single Pass Shell Fluid Velocity Variable
100	67.6	81.5	200.6	191.6	121.6	2.317	0.883	0.196	1.38	0.676	0.0347	0.0996	7,816	19,960	1.52	1.30	43.7	
101	67.6	79.0	200.1	189.3	121.4	1.117	0.848	0.196	0.80	0.393	0.0347	0.0681	7,812	13,650	1.52	0.88	35.8	
102	67.6	75.4	200.6	187.9	122.7	0.367	0.833	0.196	0.33	0.162	0.0347	0.0373	7,812	7,480	1.52	0.49	24.3	

Tube Fluid - Water
Shell Fluid - Oil "D"
Sp. Gr. - 0.892 at Average Temp.
Abs. Vis. 11.6 c.p. at Average Temp.

TABLE VIII DATA AND RESULTS WATER COOLING WATER SERIES I.

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TEMPERATURES: TUBE FLUID -OUT- DEG. F.	TEMPERATURES: SHELL FLUID -IN- DEG. F.	TEMPERATURES: SHELL FLUID -OUT- DEG. F.	TEMPERATURES: MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEFFICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	PRESSURE DROP: SHELL INCHES HG.	PRESSURE DROP: SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	FLUID FLOWING: SHELL CU. FT. PER SEC.	FLUID FLOWING: TUBE LB. PER HOUR	FLUID FLOWING: SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	FLUID VELOCITY: SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
103	86.0	101.0	124.9	115.4	26.6						0.0259	0.0419	5,812	9,402	1.14	1.09	160.5	Tests 103 to 107 incl. Double Pass Shell Fluid Velocity Variable
104	86.0	103.9	125.0	117.6	26.0						0.0256	0.0609	5,752	13,684	1.13	1.59	194.0	
105	86.6	104.9	124.3	117.9	24.9						0.0255	0.0756	5,726	16,990	1.12	1.97	206.0	
106	86.6	106.6	124.8	119.6	24.9						0.0256	0.0955	5,760	21,418	1.13	2.49	227.0	
107	86.6	108.1	125.1	120.3	24.4						0.0253	0.1132	5,682	25,444	1.11	2.94	245.0	
108	78.0	106.8	123.7	116.8	26.4						0.0104	0.0448	2,332	10,050	0.46	1.17	124.3	Tests 108 to 112 incl. Double Pass Tube Fluid Velocity Variable
109	78.0	100.0	123.7	114.0	29.3						0.0189	0.0446	4,236	10,004	0.83	1.16	155.4	
110	78.0	96.5	124.7	113.1	31.6						0.0284	0.043	6,365	9,644	1.25	1.12	181.0	
111	79.0	93.2	123.7	111.7	31.6						0.0367	0.0446	8,234	10,014	1.62	1.16	182.0	
112	79.5	92.1	123.6	111.3	31.6						0.0443	0.0440	9,940	9,873	1.95	1.15	192.0	
																		Tube Fluid - Water Shell Fluid - Water Abs. Vis. - 0.44 c.p. at Average Shell Temp.

TABLE IX DATA AND RESULTS WATER COOLING OIL "E" SERIES I

TEST NUMBER	TEMPERATURES					FLOW MANO. READING INCHES HG.	ORIFICE COEFFICIENT	PRESSURE DROP			FLUID FLOWING				FLUID VELOCITY		OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
	TUBE FLUID -IN- DEG.F.	TUBE FLUID -OUT- DEG.F.	SHELL FLUID -IN- DEG.F.	SHELL FLUID -OUT- DEG.F.	MEAN TEMP. DIFF. DEG.F.			TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	TUBE FT. PER SEC.	SHELL FT. PER SEC.		
113	84.9	88.0	124.6	120.7	35.6						0.0278	0.0448	6,246	8,718	1.23	1.17	26.7	Tests 113 to 118 incl. Double Pass Shell Fluid Vel. Variable Sp.Gr. -0.864 at Ave. Temp. Abs. Vis.- 10.4 c.p. at Average Temp.
114.	84.9	88.7	124.6	121.1	36.6						0.0277	0.0620	6,225	12,066	1.22	1.62	32.1	
115	84.9	89.4	125.3	121.9	36.6						0.0278	0.0774	6,240	15,048	1.23	2.02	37.2	
116	84.9	89.8	124.7	121.6	35.8						0.0277	0.0921	6,207	17,907	1.22	2.40	41.6	
117	84.7	90.4	125.9	122.9	36.8						0.0277	0.1083	6,207	21,054	1.22	2.82	46.8	
118	84.5	90.1	123.0	120.5	34.4						0.0279	0.1133	6,258	22,062	1.23	2.95	49.6	
119	85.1	89.2	161.1	151.1	68.9						0.0277	0.0249	6,210	4,734	1.22	0.65	18.1	Tests 119 to 124 incl. Double Pass Shell Fluid Vel. Variable Sp.Gr. -0.849 at Ave. Temp. Abs. Vis.- 5.8 c.p. at Average Temp.
120	85.3	94.1	164.8	157.0	71.3						0.0274	0.0676	6,159	12,900	1.21	1.76	37.4	
121	85.3	95.3	163.5	156.2	69.5						0.0275	0.0836	6,174	15,957	1.21	2.17	43.6	
122	85.1	96.3	163.1	156.3	69.1						0.0277	0.0988	6,216	18,834	1.22	2.57	49.4	
123	85.1	98.0	164.1	157.5	69.3						0.0270	0.1194	6,069	22,764	1.19	3.11	55.4	
124	85.1	99.3	165.2	158.7	69.9						0.0264	0.1280	5,931	24,393	1.17	3.33	58.7	Tube Fluid - Water Shell Fluid - Oil "E"

TABLE X DATA AND RESULTS WATER COOLING OIL "F" SERIES I.

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TUBE FLUID -OUT- DEG. F.	SHELL FLUID -IN- DEG. F.	SHELL FLUID -OUT- DEG. F.	MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEFFICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
125	86.9	91.9	207.2	196.3	112.2						0.0271	0.0251	6,080	4,840	1.19	0.65	13.3	Tests 125 to 132 incl. Double Pass Shell Fluid Vel. Variable Sp. Gr. - 0.859 at Ave. Temp. Abs. Vis. - 15.0 c.p. at Average Temp.
126	86.9	94.0	209.3	199.7	114.3						0.0245	0.0374	5,484	7,224	1.08	0.97	16.6	
127	86.1	95.1	208.6	200.5	113.5						0.0275	0.0529	6,164	10,248	1.21	1.38	21.2	
128	87.3	96.6	208.9	201.7	112.7						0.0273	0.0680	6,092	13,148	1.20	1.77	24.6	
129	87.3	97.5	208.4	201.6	111.2						0.0298	0.0869	6,668	16,796	1.31	2.26	29.9	
130	87.3	100.2	208.5	202.0	111.3						0.0271	0.1061	6,060	20,508	1.19	2.76	34.6	
131	87.1	101.1	208.5	202.1	110.8						0.0276	0.1193	6,168	23,056	1.21	3.11	38.1	
132	87.1	101.9	207.7	201.7	110.0						0.0273	0.1293	6,092	25,032	1.20	3.37	40.1	Tube Fluid - Water
133	86.7	97.4	278.0	253.7	173.9						0.0278	0.0271	6,180	5,080	1.22	0.71	18.7	Shell Fluid - Oil "F"
134	86.5	102.2	274.5	255.7	169.7						0.0281	0.0506	6,244	9,484	1.23	1.32	28.1	Tests 133 to 139 incl. Double Pass Shell Fluid Vel. Variable Sp. Gr. - 0.839 at Ave. Temp. Abs. Vis. - 5.6 c.p. at Average Temp.
135	86.5	107.0	278.1	260.5	171.9						0.0282	0.0720	6,268	13,508	1.24	1.87	36.4	
136	86.5	110.4	279.5	262.8	174.1						0.0282	0.0886	6,272	16,644	1.24	2.31	42.1	
137	86.4	113.5	279.8	260.5	172.5						0.0280	0.1050	6,244	19,704	1.23	2.74	48.2	
138	86.4	116.1	279.8	264.8	169.9						0.0282	0.1215	6,280	22,800	1.24	3.16	53.6	
139	86.5	120.4	280.3	266.8	170.2						0.0282	0.1448	6,280	27,152	1.24	3.77	61.0	
140	86.9	89.1	152.8	149.4	62.6						0.0253	0.0231	5,880	4,548	1.11	0.60	9.9	Tests 140 to 145 incl. Double Pass Shell Fluid Vel. Variable Sp. Gr. - 0.882 at Ave. Temp. Abs. Vis. - 41.0 c.p. at Average Temp.
141	86.8	90.0	150.4	147.1	60.4						0.0266	0.0479	5,952	9,405	1.17	1.25	15.7	
142	86.7	90.9	151.9	149.0	61.4						0.0268	0.0669	5,997	13,167	1.18	1.74	19.8	
143	86.7	91.6	150.9	148.3	60.3						0.0268	0.0872	5,991	17,160	1.18	2.27	23.6	
144	86.7	91.7	150.5	147.9	60.2						0.0271	0.0937	6,031	18,417	1.19	2.44	24.3	
145	86.7	92.3	150.6	148.1	59.9						0.0270	0.1110	6,030	21,795	1.19	2.89	27.5	
146	86.9	99.6	199.2	191.7	101.9						0.0152	0.0616	3,384	11,916	0.67	1.61	20.4	Tests 146 to 150 incl. Double Pass Tube Fluid Vel. Variable Sp. Gr. - 0.868 at Ave. Temp. Abs. Vis. - 32.0 c.p. at Average Temp.
147	87.3	94.5	200.4	192.2	105.3						0.0278	0.0620	6,212	12,008	1.22	1.62	21.1	
148	87.1	92.3	200.5	193.0	106.5						0.0405	0.0617	9,056	11,944	1.78	1.61	21.7	
149	87.1	91.1	200.3	192.6	107.3						0.0530	0.0611	11,864	11,828	2.33	1.59	21.7	
150	87.0	90.3	200.0	192.0	109.9						0.0659	0.0618	14,728	11,968	2.90	1.61	21.8	
151	86.7	91.7	201.1	192.8	107.7						0.0278	0.0324	6,200	6,264	1.22	0.42	14.1	Tests 151 to 155 incl. Single Pass Shell Fluid Vel. Variable Sp. Gr. - 0.868 at Ave. Temp. Abs. Vis. - 32.0 c.p. at Average Temp.
152	86.7	95.0	198.2	192.1	104.3						0.0275	0.0545	6,160	10,548	1.21	0.71	23.9	
153	86.7	97.9	199.3	193.3	104.0						0.0247	0.0759	5,532	14,696	1.08	0.99	29.0	
154	86.7	98.6	201.3	195.8	106.0						0.0278	0.0946	6,204	18,300	1.22	1.23	33.6	
155	86.8	100.1	201.5	197.2	105.2						0.0280	0.1140	6,256	22,040	1.23	1.48	38.5	

TABLE XI DATA AND RESULTS WATER COOLING OIL "G" SERIES 1

TEST NUMBER	TEMPERATURES: TUBE FLUID -IN- DEG. F.	TUBE FLUID -OUT- DEG. F.	SHELL FLUID -IN- DEG. F.	SHELL FLUID -OUT- DEG. F.	MEAN TEMP. DIFF. DEG. F.	FLOW MANO. READING INCHES HG.	ORIFICE COEFFICIENT	PRESSURE DROP: TUBE LB. PER SQ. IN.	SHELL INCHES HG.	SHELL LB. PER SQ. IN.	FLUID FLOWING: TUBE CU. FT. PER SEC.	SHELL CU. FT. PER SEC.	TUBE LB. PER HOUR	SHELL LB. PER HOUR	FLUID VELOCITY: TUBE FT. PER SEC.	SHELL FT. PER SEC.	OVERALL TRANSFER COEFFICIENT B.T.U./HR./SQ.FT. PER °F. M.T.D.	REMARKS
156	85.8	110.0	153.1	143.8	50.0						0.0124	0.0833	2,772	14,724	0.55	2.17	66.3	Tests 156 to 161 incl. Double Pass Tube Fluid Vel. Variable
157	85.6	99.6	152.3	139.4	53.3						0.0288	0.0830	6,477	14,697	1.27	2.16	82.9	
158	85.7	96.3	152.2	138.6	54.4						0.0411	0.0827	9,222	14,613	1.81	2.15	82.8	
159	86.0	94.2	153.1	138.9	55.9						0.0540	0.0764	12,156	13,506	2.38	1.99	91.0	
160	86.0	93.0	152.5	137.8	55.6						0.0670	0.0813	15,027	14,391	2.95	2.12	93.5	
161	85.8	91.7	152.7	137.7	56.3						0.0850	0.0826	18,915	14,610	3.75	2.15	97.5	
162	85.6	96.5	152.0	136.8	52.3						0.0272	0.0503	6,099	8,895	1.20	1.31	61.9	Tests 162 to 167 incl. Double Pass Shell Fluid Vel. Variable
163	85.6	98.2	150.8	137.8	52.4						0.0279	0.0685	6,252	12,120	1.23	1.78	73.5	
164	85.6	100.0	151.3	139.5	52.6						0.0278	0.0851	6,207	15,048	1.22	2.22	82.3	
165	85.6	101.4	151.3	140.5	52.2						0.0279	0.1009	6,246	17,841	1.23	2.62	92.0	
166	85.7	103.1	150.9	141.4	51.4						0.0277	0.1264	6,198	22,347	1.22	3.29	102.5	
167	85.6	103.4	152.0	142.3	51.8						0.0281	0.1322	6,279	23,388	1.24	3.44	105.5	
168	85.7	104.7	149.8	139.8	49.4						0.0276	0.1270	6,195	22,449	1.21	1.65	116.5	Tests 168 to 173 incl. Single Pass Shell Fluid Vel. Variable
169	85.8	103.1	150.7	139.2	51.5						0.0276	0.1072	6,198	18,975	1.21	1.40	106.8	
170	85.6	100.9	150.9	138.1	52.1						0.0281	0.0869	6,291	15,366	1.24	1.13	92.5	
171	85.3	99.7	150.3	136.8	51.0						0.0261	0.0694	5,835	12,288	1.15	0.91	80.8	
172	85.1	97.1	150.0	135.4	51.1						0.0266	0.0506	5,949	8,949	1.17	0.61	68.6	
173	84.7	94.9	151.9	132.7	52.2						0.0273	0.0348	6,087	6,144	1.20	0.45	58.1	Tube Fluid - Water Shell Fluid - Oil "G" Sp.Gr. -0.785 at Ave. Temp. Abs. Vis. -0.9 c.p. at Average Temp.

IV. DISCUSSION OF RESULTS

Mechanism of heat transmission.—In open vessels where a gas or a vapor separates the warmer substance from the cooler substance heat is transmitted largely by radiation; but in vessels where the heat must pass through a liquid or a solid, the kind of transmission is principally that of conduction.

If the fluids were quiescent, heat would be transmitted by conduction alone and the rate of heat transmission would be dependent upon the conductivity and the thickness of the substance. If, however, the fluids are in motion, as is the case in a commercial heat exchanger, heat will flow not only by conduction but also by convection.

The particles of the fluid do not move at the same rate through the vessel. That portion of the stream nearest the wall moves at a slower rate than the central part. The variation of the velocity across a section of the path is governed by the viscosity of the fluid and the width of the path. The thickness of the layer of the slow-moving particles near the wall of the vessel is also governed by the viscosity of the fluid and the width of the path.

Since the heat transferred in an exchanger is principally by conduction and convection, the rate at the point of transfer will be governed by the thickness of the slow-moving film at the surface of the fluid. The rate at which the heat is conveyed to the point of transfer will be controlled by the average velocity of the stream, its density, and its heat capacity. Also, the total amount of heat transferred will be governed by the relation of the width of the path to its length.

From a summary of the foregoing statements it may be noted that the transfer of heat in the case of a moving fluid in a closed vessel is governed by the diameter (width) of the vessel, the length of the path, the heat capacity (specific heat) of the fluid, its conductivity, density (specific gravity), velocity, and viscosity.

Transfer relation developed by Morris and Whitman.—Morris and Whitman[3] have presented data on a single tube experimental heat exchanger which shows that a relation exists between the groups of the variables $\frac{hD}{K}$, $\frac{CZ}{K}$, and $\frac{DV}{Z}$.

That is, $\dfrac{\frac{hD}{K}}{\left(\frac{CZ}{K}\right)^{0.37}}$ is a function of $\frac{DV}{Z}$

where

h = film coefficient
D = diameter of the containing vessel in inches
K = conductivity of the fluid in B.t.u. per hour per square foot per foot of thickness per degree Fahrenheit m.t.d.
C = specific heat of the fluid
Z = absolute viscosity in centipoises
V = mass velocity in pound per second per square foot of cross-sectional area.

These men, however, did not consider the relation of the length of the path to its diameter (width). A replot of their data is shown by Fig. 5.

Transfer relation developed by McAdams and Frost.—Messrs. McAdams and Frost[4] have presented data that show the relation of the rate of heat transfer to all of the variables given above.

A replot of their data with interpolation and extension for different values of $\frac{CZ}{K}$ to cover the range encountered in these tests is shown by Fig. 6.

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[3]Morris and Whitman, "Heat Transfer for Oils and Water in Pipes," Industrial and Engineering Chemistry, Vol. 20, p. 232.

[4]McAdams and Frost, "Heat Transfer," *Journal of Industrial and Engineering Chemistry,* Vol. 14, No. 1, p. 13.

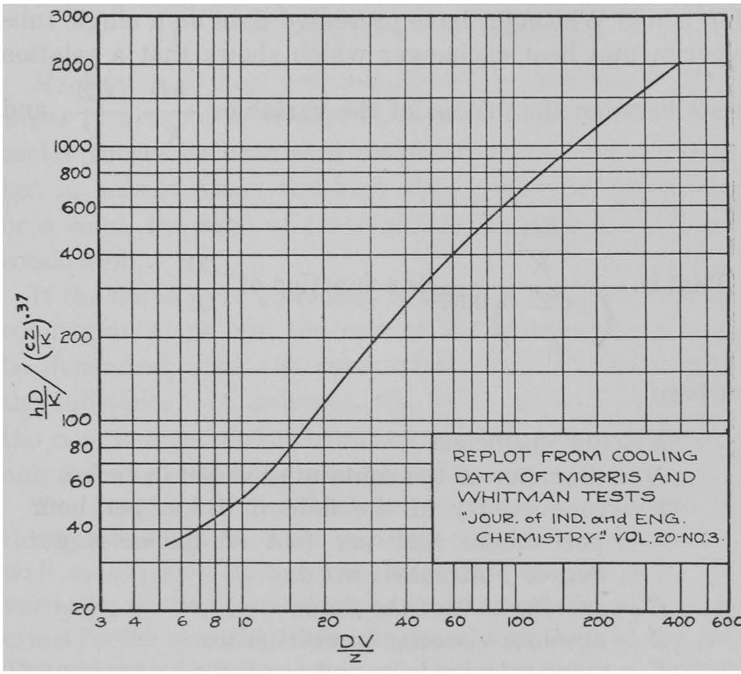

FIG. 5. REPLOT FROM COOLING DATA OF MORRIS AND WHITMAN TESTS

These curves show that for a given value of $\frac{CZ}{K}$,

$$\frac{\frac{hD}{K}}{\left(1+\frac{50}{r}\right)} \text{ is a function of } \frac{DV}{Z} \qquad (5)$$

where

r = ratio of the length of the pass to its diameter.

(The other symbols have the same meaning as given above for the Morris and Whitman data.)

The group $\frac{CZ}{K}$ shows the ratio of the product of the specific heat and absolute viscosity (in centipoises) to the conductivity of the fluid in B.t.u. per square foot per hour per foot of thickness per degree Fahrenheit mean temperature difference.

Both Morris and Whitman's and McAdams and Frost's data point to a relation existing between the overall heat transfer coefficient and $\frac{DUS}{Z}$. Since the diameter (D) is constant for a given vessel, there exists a relation between the linear velocity of the fluid (U), the specific gravity (S), and the absolute viscosity (Z) with the overall heat transfer coefficient. Such a relation is shown by Fig. 7.

Film coefficients versus overall coefficients.—Overall transfer coefficients computed for the fluids under the conditions of these tests from the data of McAdams and Frost were found to vary approximately 5 per cent from the average values found experimentally.

As an example:

Oil "A" (Shell Fluid) flowing in *Double Pass*
Water (Tube Fluid) as cooling medium.
(Data taken from test number 29.)

Shell Side

$$\frac{CZ}{K} = \frac{0.51 \times 0.9}{0.078} = 5.9$$

$$\frac{DV}{Z} = \frac{3.59 \times 5.59 \times 62.4 \times 0.804}{0.9} = 1117$$

From the curve, Fig. 6, the corresponding coördinate G at $\frac{DV}{Z} = 1117$ and $\frac{CZ}{K} = 5.9$ equals 5300 and

$$G = \frac{\frac{h_s D}{K}}{1 + \frac{50}{r}} = \frac{\frac{3.59 h_s}{0.078}}{1 + \frac{50}{10 \times 12}} = 5300$$

$$h_s = \frac{5300 \times 2.495}{46} = 288 \text{ B.t.u. per hour per sq. ft. per degree Fahr. m.t.d.}$$

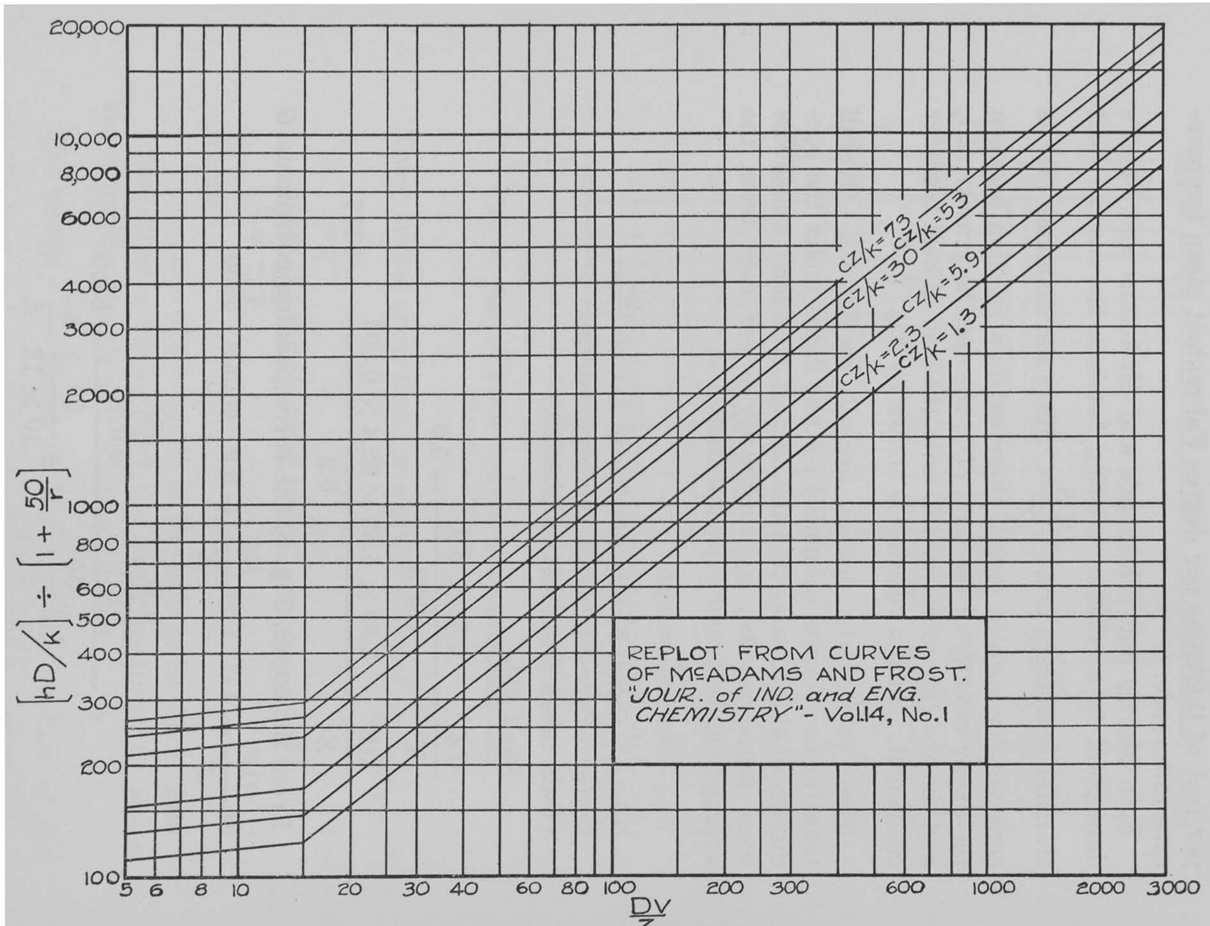

FIG. 6. REPLOT FROM CURVES OF MCADAMS AND FROST

Tube Side

$$\frac{CZ}{K} = \frac{1 \times 0.96}{0.37} = 2.6$$

$$\frac{DV}{Z} = \frac{0.527 \times 1.53 \times 62.4 \times 1}{0.96} = 52.4$$

From the curve, Fig. 6,

$$G = 390 \text{ at } \frac{DV}{Z} = 52.4 \text{ and } \frac{CZ}{K} = 2.6$$

or

$$G = \frac{\dfrac{h_t D}{K}}{1 + \dfrac{50}{r}} = \frac{\dfrac{0.527 h_t}{0.37}}{1 + \dfrac{50}{\dfrac{10 \times 12}{0.527}}} = \frac{1.505 h_t}{1.22}$$

$h = 316$ B.t.u. per hour per square foot per degree Fahrenheit m.t.d.

The overall transfer coefficient U then becomes equal to

$$\frac{1}{\dfrac{1}{h_s} + \dfrac{L}{h_o} + \dfrac{1}{h_t}} \qquad (6)$$

where

h_s = film coefficient on the shell side
h_t = film coefficient on the tube side
h_o = conductivity of the tube metal in B.t.u. per hour per square foot per inch of thickness per degree Fahrenheit m.t.d.
L = thickness of tube wall in inches.

$$U = \frac{1}{\dfrac{1}{288} + \dfrac{0.049}{660} + \dfrac{1}{316}} = 149 \text{ B.t.u. per hour per square foot per degree Fahr. m.t.d.}$$

The experimental coefficient for the same shell and tube velocities was 145 B.t.u. per square foot per hour per degree Fahrenheit mean temperature difference.

A second example:

Oil "A" (Shell Fluid) flowing in *Single Pass*
Water (Tube Fluid) as cooling medium.
(Data taken from test number 39.)

Shell Side

$$\frac{CZ}{K} = \frac{0.51 \times 0.9}{0.078} = 5.9$$

$$\frac{DV}{Z} = \frac{5.07 \times 3.02 \times 62.4 \times 0.804}{0.078} = 853$$

From the curve, Fig. 6,

$$G = 4220 \text{ at } \frac{DV}{Z} = 853 \text{ and } \frac{CZ}{K} = 5.9$$

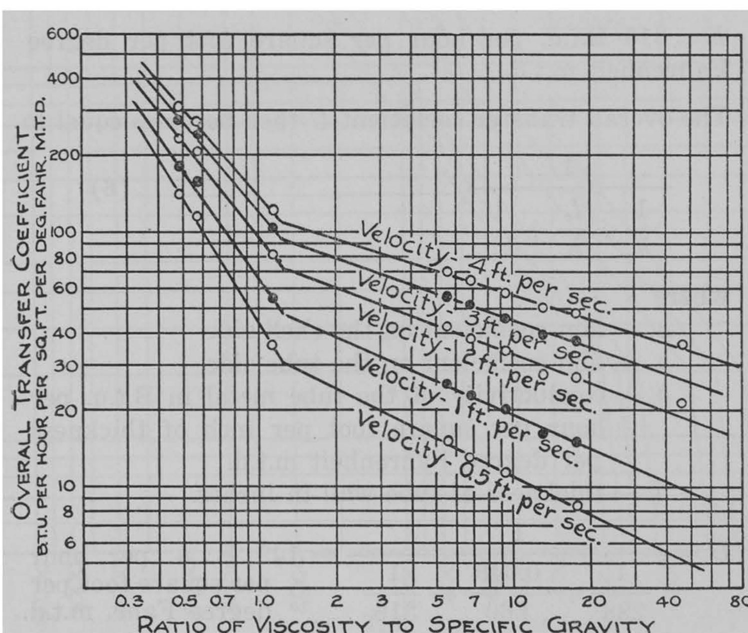

FIG. 7. RELATION OF OVERALL TRANSFER COEFFICIENT TO RATIO OF VISCOSITY TO SPECIFIC GRAVITY

$$G = \frac{\dfrac{h_s D}{K}}{1 + \dfrac{50}{r}} = \frac{\dfrac{5.07}{0.078} h_s}{1 + \dfrac{50}{\dfrac{5 \times 12}{5.07}}} = \frac{65}{5.22} h_s = 4220$$

$h_s = 339$ B.t.u. per square foot per hour per degree Fahrenheit m.t.d.

Tube Side

$$\frac{CZ}{K} = \frac{1 \times 0.96}{0.37} = 2.6$$

$$\frac{DV}{Z} = \frac{0.527 \times 1.54 \times 62.4 \times 1}{0.94} = 53.8$$

From the curve, Fig. 6,

$$G = 400 \text{ at } \frac{DV}{Z} = 53.8 \text{ and } \frac{CZ}{K} = 2.6$$

$$G = \frac{\dfrac{h_t D}{K}}{1 + \dfrac{50}{r}} = \frac{\dfrac{0.527}{0.37} h_t}{1 + \dfrac{50}{\dfrac{10 \times 12}{0.529}}} = 400$$

$h_t = 325$ B.t.u. per square foot per hour per degree Fahr. m.t.d.

Then $U = \dfrac{1}{\dfrac{1}{339} + \dfrac{0.049}{660} + \dfrac{1}{325}} = 164$ B.t.u. per hour per sq. ft. per degree Fahr. m.t.d.

The experimental coefficient as calculated from test number 39 was found to be 163 B.t.u. per hour per square foot per degree Fahrenheit mean temperature difference.

Film transfer coefficient equation.—The equation

$$h = 374\left(1 + \frac{50}{r}\right)\frac{C^{0.2}\,K^{0.8}\,S^{0.784}\,v^{0.784}}{D^{0.216}\,Z^{0.584}} \qquad (7)$$

is satisfied, approximately, by the experimental coefficients. In this equation the symbols are:

h = film transfer coefficient on one side of the container in B.t.u. per square foot per hour per degree Fahrenheit m.t.d.
r = ratio of the length of path to its diameter
K = conductivity coefficient of the fluid in B.t.u. per square foot per hour per foot of thickness per degree Fahr. m.t.d.
C = specific heat
v = linear velocity in feet per second
S = specific gravity
Z = absolute viscosity in centipoises
D = diameter of the vessel in inches.

Applying the above equation to the data of test number 29, the calculations are as follows:

Oil "A" (Shell Fluid) flowing in *Double Pass*
Water (Tube Fluid) as cooling medium.

The Shell Fluid coefficient would be

$$h_s = 374\left(1 + \frac{\dfrac{50}{2\times 5}}{\dfrac{3.59}{12}}\right)\frac{(0.51)^{0.2}\,(0.078)^{0.8}\,(0.804)^{0.784}\,(5.59)^{0.784}}{(3.59)^{0.216}\,(0.9)^{0.584}}$$

h_s = 277 B.t.u. per hour per square foot per degree Fahr. m.t.d.

and the tube fluid coefficient would be

$$h_t = 374\left(1 + \frac{\dfrac{50}{2\times 5}}{\dfrac{0.527}{12}}\right)\frac{(1.0)^{0.2}\,(0.36)^{0.8}\,(1)^{0.784}\,(1.53)^{0.784}}{(0.527)^{0.216}\,(1)^{0.584}}$$

h_t = 323 B.t.u. per hour per square foot per degree Fahr. m.t.d.

The overall coefficient then becomes

$$U = \frac{1}{\frac{1}{277} + \frac{0.049}{660} + \frac{1}{323}} = 147.6 \text{ B.t.u. per hour per square foot per degree Fahr. m.t.d.}$$

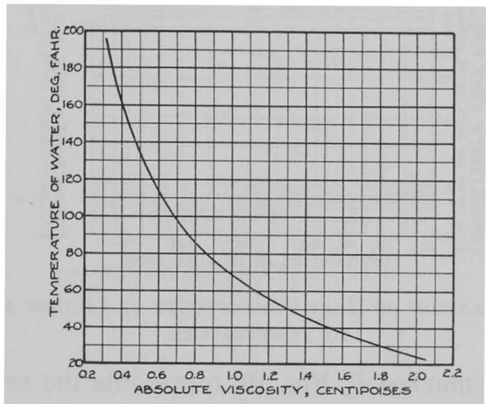

FIG. 8. RELATION OF WATER VISCOSITY TO TEMPERATURE

The experimental value for this test was 145 B.t.u. per hour per square foot per degree Fahrenheit mean temperature difference.

At stated before, the values of specific heat were calculated from Fortsch and Whitman's equation. The conductivity of water at various temperatures is shown by Fig. 8, and the conductivities of oils and that of brass at different temperatures are given by the curves in Fig. 9.

Diameter of shell passage.—In making a comparison with the data of Morris and Whitman and with that of McAdams and Frost, it was necessary to set a value for the diameter of the passage through which the fluid moved during the heat exchanging process. The true equivalent diameter of a passage of this kind is questionable, for the hydraulic relations would depend upon the *wetted perimeter* of the pass, while the conduction of heat would depend upon the radial distance from the cool surface to the point from whence the heat comes. Heat will pass from any point to another if there is a difference in temperature, and the amount of heat flow will depend upon the resistance of the path and the thermal head.

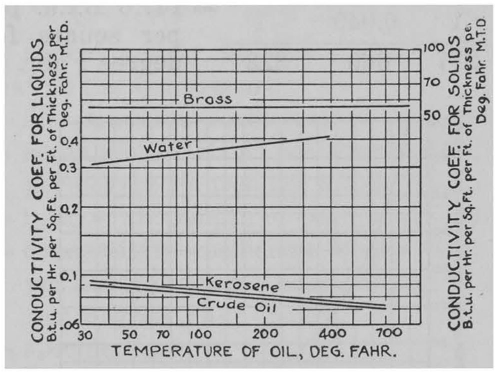

FIG. 9. RELATION OF HEAT CONDUCTIVITY OF LIQUIDS AND SOLIDS TO TEMPERATURE

The circle marked *B*, Fig. 10, represents the area equivalent to the gross area of one-half of the exchanger cross-section. The diameter of this circle was used as the diameter of the shell passage in the double-pass "set up" for all of the calculations involving this factor. The diameter of a circle of twice the area of circle *B* was used for the single-pass "set-up."

The external circle marked *A* represents the equivalent perimeter equal to the *wetted perimeter* of the tubes in one pass (fifteen tubes) of the exchanger. The circle marked *C* represents an area equal to the shell passage area of one pass divided by the number of tubes in that pass.

It appears that the heat conducted from the fluid in the shell to the tube would be governed by a diameter approximately equal to that of circle *C*. That is, that the variation of the velocity across the passage would be controlled by the "free" space between the tubes; but, when it is considered that the shell of the exchanger is the only true containing wall, it should be seen that the diameter of the shell area would be more nearly that with which the velocity would vary. Then, too, some heat is probably conducted from the remote portions of the passage to the centrally

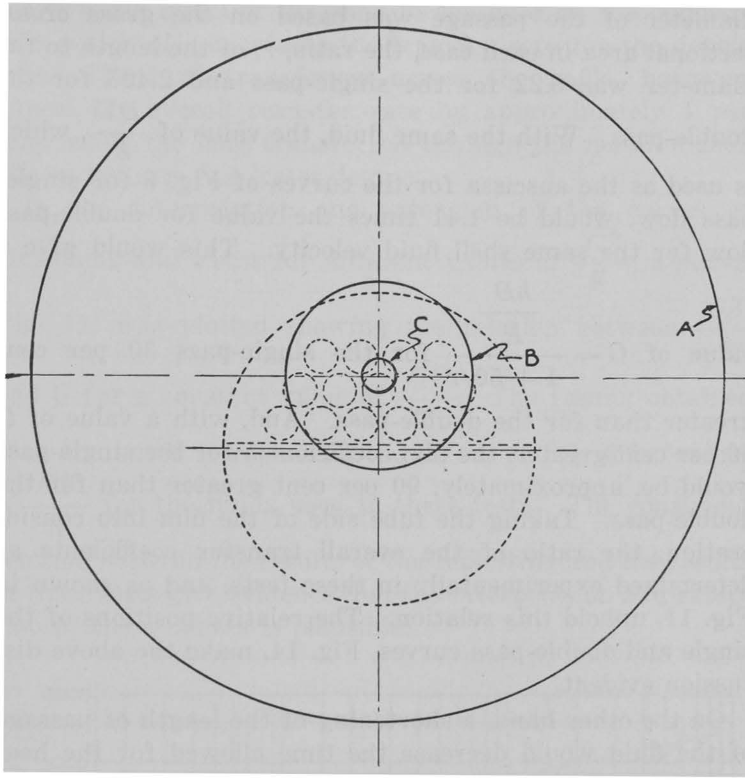

FIG. 10. COMPARATIVE SHELL PASSAGE CROSS-SECTIONAL AREAS

located tubes, which would infer that the diameter as used would be more nearly the true equivalent diameter.

Effect of length of path of flow.—The factor $\left(1 + \frac{50}{r}\right)$ as given by McAdams and Frost in their equation for heat transfer across a film of fluid, is substantiated by the results of these tests, since there was a greater overall heat transfer rate shown by the single-pass flow than was shown by the double-pass flow for the same oil flowing at the same velocity and entering the exchanger at the same temperature. The length of the path for single-pass flow was one-half of the length for double-pass flow; and, since the diameter of the passage was based on the gross cross-sectional area in each case, the ratio, r, of the length to the diameter was 5.22 for the single-pass and 2.495 for the double-pass. With the same fluid, the value of $\frac{DV}{Z}$, which is used as the abscissa for the curves of Fig. 6 for single-pass flow, would be 1.41 times the value for double-pass flow for the same shell fluid velocity. This would give a value of $G = \dfrac{\frac{hD}{K}}{1 + 50/r}$ for the single-pass 30 per cent greater than for the double-pass. And, with a value of G 30 per cent greater, the film coefficient h for the single-pass would be, approximately, 90 per cent greater than for the double-pass. Taking the tube side of the film into consideration, the ratio of the overall transfer coefficients as determined experimentally in these tests, and as shown in Fig. 11, uphold this relation. The relative positions of the single and double-pass curves, Fig. 14, make the above discussion evident.

On the other hand, a shortening of the length of passage of the fluid would decrease the time allowed for the heat exchange. This would indicate that the time necessary for the heat exchange to take place between the hot and cold fluids is less than the time necessary for the fluid to pass the length of the double-pass and that the efficiency of heat transfer is reduced because of too long a path of flow.

The lack of consistency in the relation of the experimental results with that as calculated using Morris and Whitman's data may be attributed, principally, to the omission of the length of the pass from their data.

Effect of heat short-circuiting through metal of shell and baffle.—Some heat will pass from the fluid in the shell on one side of the central baffle across the baffle to the fluid on the opposite side, thereby decreasing the overall transfer rate in the case of the double-pass flow. And, too, some heat will pass from the fluid on one side of the baffle through the metal of the shell to the fluid on the opposite side, which also decreases the transfer rate for the double pass. The heat transferred across the baffle, however, affects the overall transfer rate by approximately 1 per cent, while the heat transferred through the metallic shell affects the rate to a lesser degree.

In the interpolation and extension of the curves of McAdams and Frost for different values of $\frac{CZ}{K}$, a curve, Fig. 11, was plotted showing the relation between $\frac{CZ}{K}$ and G for a constant value of $\frac{DV}{Z}$. The results obtained were then used in replotting the curves of Fig. 6 for values of $\frac{CZ}{K}$ for the fluids used in these tests. The consistent relation between the results of the replotting and the results of these tests give evidence that the interpolation and extension of these curves is plausible.

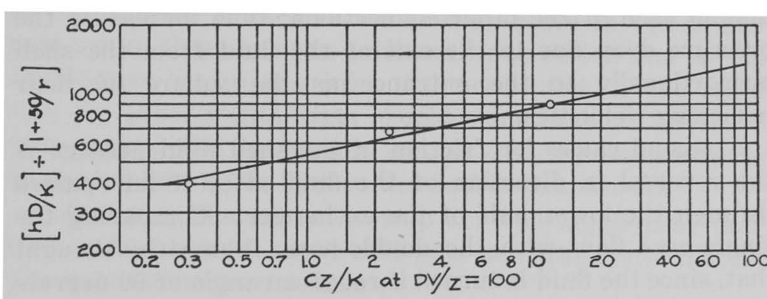

FIG. 11. RELATION OF $\frac{CK}{K}$ AND G FOR A CONSTANT VALUE OF $\frac{DV}{Z}$

Factors affecting fluid friction.—In considering the pressure drop through the exchanger it is necessary to refer to the general layout of the apparatus, as shown in Fig. 2, to justify the results that were obtained.

It is considered in hydraulics that the drop in pressure caused by the friction of the path of a fluid through a vessel is proportional to the square of the velocity. Such a proportionality does exist in a symmetrical vessel, but it is doubtful whether this theory can be applied to a container of the type used in these tests without considering the individual losses that cause a reduction in pressure.

For the double-pass set-up the fluid enters the bottom half of the shell at one end, passes the length of the tubes, is reversed in direction, and then returns through the upper half to the outlet on the same end as the inlet. Since the inlet and outlet connections are of the same size, the velocity will be the same at these points and, consequently, the pressure drop due to the sudden enlargement and contraction would be the same.

When the single-pass set-up is used, the fluid enters both sides of the shell at one end; that is, it enters through both the inlet and outlet connections for the double pass. In this case it leaves the shell at the opposite end after having passed the length of the exchanger. Since the outlet connection for the single pass is the same size as each of the other connections, the velocity in the outlet will be twice that in each of the other connections; thus increasing the pressure drop due to the exit of the fluid from the shell proportionally to the entrance as the square of their respective velocities.

A second cause for friction in the shell fluid passage is the reversal in direction of the fluid after it has passed through the lower half of the exchanger. Comparing the single-pass flow with the double-pass, it may be thought that, since the fluid is turned through an angle of 90 degrees in the single-pass set-up, while it is completely reversed in direction in the other type of set-up, the resulting friction due to flow would be less in the first case than in the second. To assume that the loss produced at this point is as large in one case as in the other is not likely to be appreciably in error, because the two parallel streams join at this point in the case of the single-pass flow and, as a result, additional turbulence, which is not present in the double-pass set-up, is produced.

The third cause for a loss in pressure by the fluid flowing through the exchanger is fluid *pipe friction.* In long pipes and similar containers this item is usually the largest; but in this, a short vessel, it is much smaller comparatively than the other losses which have been considered. This loss in the single-pass set-up would be, approximately, 40 per cent of the same loss in the double-pass set-up.

Fluid friction in single pass versus fluid friction in double pass.—If, then, it is assumed that the pressure loss due to reversal of direction in the shell is the same in each type of set-up, and that the pipe friction is comparatively negligible, the total drop across the exchanger in the case of the single-pass flow will be approximately 230 per cent of the total drop for the double-pass flow for the same shell fluid velocity. That is, there are two losses at entry in the single-pass, each of which is equal to the loss at entry or exit in the double pass, and in the single-pass flow there is a loss at exit which is 400 per cent of each of the losses at entry because the velocity in this outlet is twice that of the velocity in each of the inlets. Since in the single-pass set-up there are two inlets and one outlet, and in the double-pass set-up one inlet and one outlet of the same size, the experimental data should show that the pressure drop across the exchanger in the single-pass set-up is approximately twice that in the double-pass set-up.

Calculating the pressure drop for the single-pass flow from data taken for the double-pass flow, the above discussion may be made more clear.

Example for Oil "D" at a velocity of 2 feet per second:

$$\frac{DVS}{Z} = \frac{3.59 \times 2 \times 0.892}{11.6} = 0.551$$

From standard pipe friction curves, $f = 0.0114$.

From the standard equation for pressure drop in straight pipes,

$$p = \frac{0.323\, fLSv^2}{D} \qquad (8)$$

where

p = pressure drop in pounds per square inch
f = friction factor
L = length of pass in feet
S = specific gravity of fluid
v = velocity of fluid in feet per second
D = diameter of container in inches
Z = absolute viscosity in centipoises.

$$p = \frac{0.323 \times 0.0114 \times 10 \times 0.892 \times (2)^2}{3.59}$$

$$= 0.0366 \text{ lb. per square inch.}$$

The remaining pressure drop is then divided equally among the losses due to entry, exit, and reversal of direction, or, each is equal to

$$\frac{0.68 - 0.0366}{3} = 0.1878 \text{ lb. per square inch.}$$

0.68 may be found to be the total pressure drop across the shell fluid, Oil "D" flowing in double pass at a velocity of 2 feet per second, Fig. 12.

The total losses for the single-pass flow should then be

Shell friction (0.0366) 0.4	= 0.0146
Reversal of flow	= 0.1878
Two entries = 2 $\times$ 0.1878	= 0.3756
Exit = 0.1878 $\left(\frac{4}{2}\right)^2$	= 0.7512
Total	= 1.3292

The curve for Oil "D," Fig. 12, for single-pass flow at a velocity of two feet per second shows a pressure drop of 1.32 pounds per square inch.

Pressure-drop equations.—An analysis of the curves for all of the oils shown by Fig. 13 reveals two equations which are satisfied, approximately, by the data from which these curves were plotted.

For single-pass flow,

$$p = \frac{0.3\ v^{1.5}}{\left(\frac{S}{Z}\right)^{0.159}} \qquad (9)$$

while for the double-pass flow,

$$p = \frac{0.15\ v^{1.5}}{\left(\frac{S}{Z}\right)^{0.159}} \qquad (10)$$

where the symbols have the same meaning as given for equation (8).

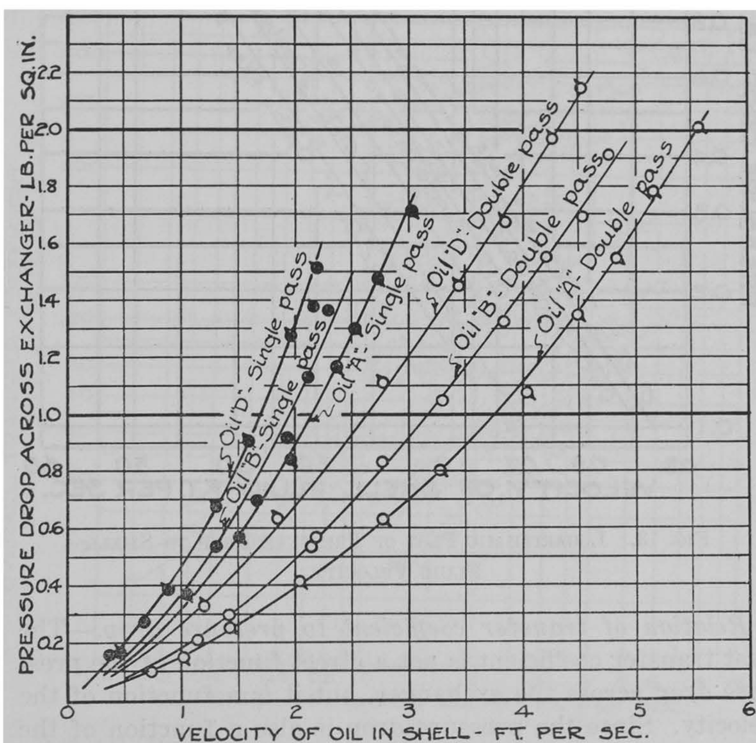

FIG. 12. RELATION OF PRESSURE DROP TO SHELL FLUID VELOCITY

Equations (9) and (10) show the pressure drop varying as the three-halves power of the linear shell velocity instead of with the square of the velocity. Since the total drop is given in the data, the above relation between the pressure drop and velocity is justified, due to the different factors that make up the total pressure drop.

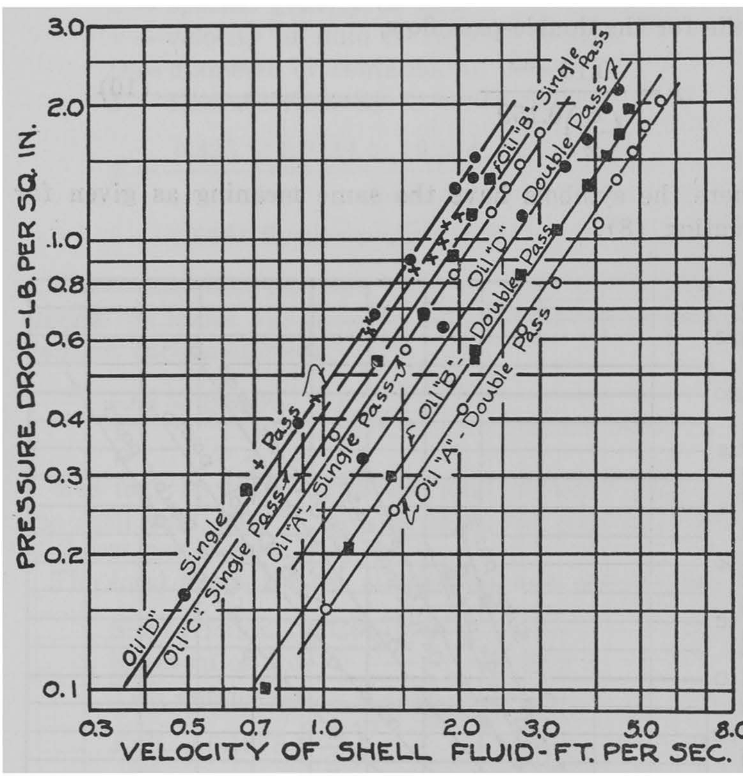

FIG. 13. LOGARITHMIC PLOT OF PRESSURE DROP TO SHELL FLUID VELOCITY

Relation of transfer coefficient to pressure drop.—The heat transfer coefficient is not a direct function of the pressure drop across the exchanger, but it is a function of the velocity. Since the pressure drop is also a function of the velocity, the coefficient may be plotted as a function of the pressure drop. Because the velocity of the water which passed through the tubes was maintained approximately constant, the overall transfer rate was plotted against the shell fluid velocity and, also, against the pressure drop across the shell. The curves are shown by Figs. 14 and 15, respectively.

The design of heat-exchanging apparatus should be such that the pipe friction losses are a minimum and the transfer rate a maximum. The pressure drop through the various types of exchangers will vary widely with different designs. Therefore, it is not considered that the data for the pressure drop nor the equations derived from the data will hold for any other type of exchanger. The data show, however, that the principal losses with baffling parallel to the tubes

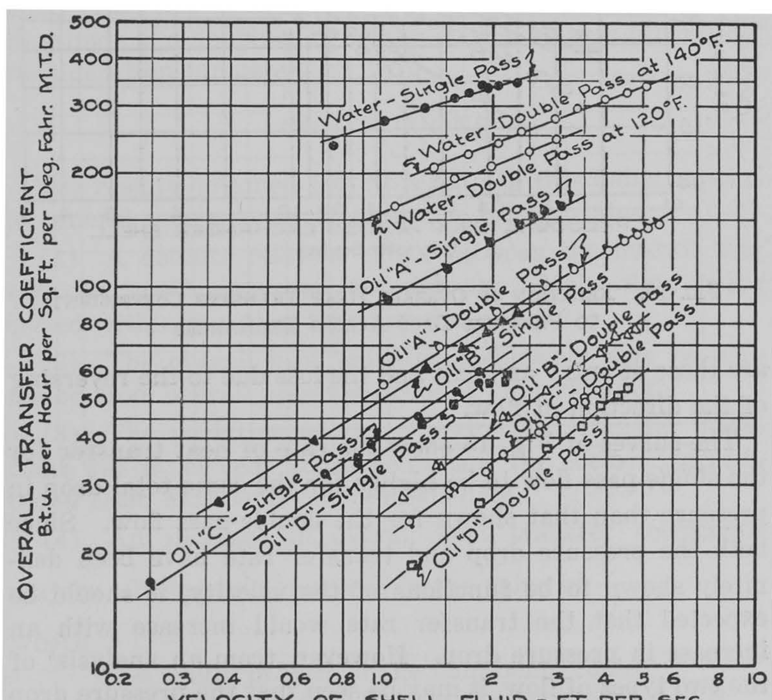

FIG. 14. RELATION OF OVERALL HEAT TRANSFER COEFFICIENT TO SHELL FLUID VELOCITY

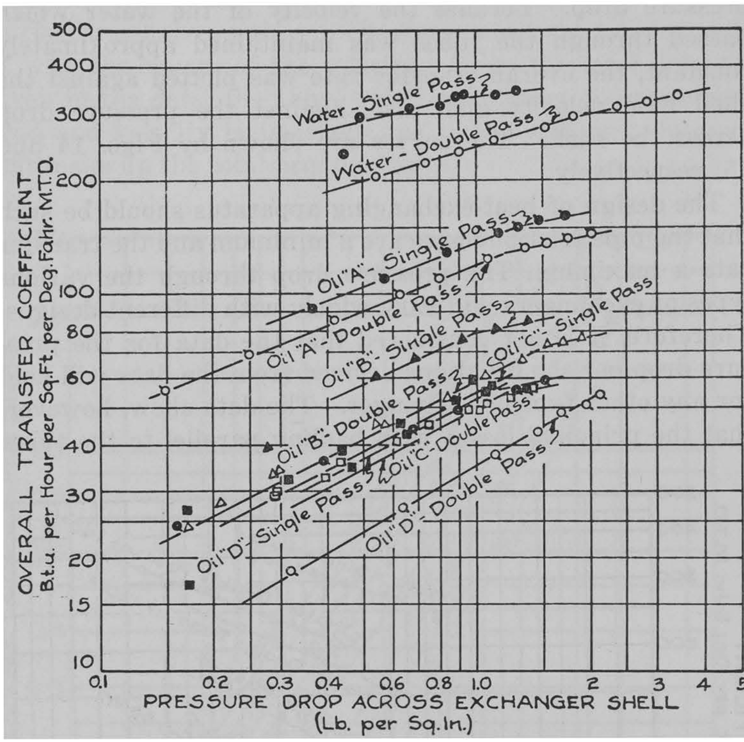

FIG. 15. RELATION OF OVERALL HEAT TRANSFER COEFFICIENT TO PRESSURE DROP ACROSS EXCHANGER

are those of entry and exit and the loss due to the reversing of the direction of flow.

The curves of Fig. 15 show the rate of heat transfer for the single-pass flow to be higher for the same total drop in pressure than that shown for the double-pass flow. Since both the pressure drop and transfer rate have been definitely shown to be functions of the velocity, it should be expected that the transfer rate would increase with an increase in pressure drop. However, from an analysis[5] of the two types of flow, it may be seen that the pressure drop over the transfer path of the single-pass flow would be

[5]See example on pressure drops on page 41.

approximately 40 per cent of that of the double pass for the same linear velocity of the shell fluid.

It has already been stated that the film transfer rate for the single pass should be, approximately, 90 per cent greater than that of the double pass for an equal shell velocity, which, if considered in conjunction with the pressure drop along the transfer surface, would indicate that the overall transfer rate for the single pass would be five times that of the double pass for the same pressure drop.

In this particular set-up, the pressure drop across the entire exchanger was nearly 100 per cent greater in the case of the single-pass flow than it was in the case of the double-pass flow for the same linear velocity of the shell fluid. For this reason there is not the wide divergence in the single and double-pass curves of Fig. 15 that theoretical consideration indicates.

CONCLUSIONS

The results obtained from the data in this bulletin point to general conclusions which may be briefly stated:

(1) A definite relation exists between the overall heat transfer coefficients and the velocity of the fluid being cooled within practical limits of the velocity.

(2) The overall heat transfer coefficients may be estimated from overall pressure drops of an exchanger.

(3) The variation of the experimental transfer coefficient was 5 per cent above and below the coefficients computed from McAdams' and Frost's data.

(4) An increase in overall heat transfer coefficients may be effected by a decrease in the value of $\frac{L}{D}$

where

L = length of pass of the fluid being cooled
D = diameter of the pass of the fluid being cooled.

(5) By replotting McAdams' and Frost's heat transfer data with interpolation and extension for different values

of $\frac{CZ}{K}$ to cover the range of fluid used, the overall heat transfer coefficient of any exchanger may be estimated to a fair degree of accuracy.

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